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COMPARATIVE ANALYSIS OF VIRTUAL AND EXPERIMENTAL PROVING GROUND FOR MEDIUM-DUTY TACTICAL TRUCK USING VARIETY OF SUSPENSIONS

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ABSTRACT

This paper presents the comparative analysis of virtual and experimental proving ground for the performance capabilities of front suspensions in the Family of Medium Tactical Vehicles (FMTV) cargo truck. The front suspension of the current baseline FMTV is a solid axle with leaf springs and shock absorbers. Two other types of suspensions including passive and semi-active suspensions are evaluated in solid and fully independent axle configurations. Virtual proving ground for on- and off-road tests are simulated in the Trucksim environment to include constant radius circular steer, double lane change, sinusoidal steer, washboard road surfaces, and half-round curb strike. Physical proving ground tests are conducted to provide some experimental correlation and validation of the baseline vehicle simulation results. The comprehensive experiments also evaluate the capabilities of various suspensions which have been considered in future FMTV design for mobility performance improvement.

INTRODUCTION

Based on the mechanical design and configuration, suspensions are generally categorized in two main groups: solid axle suspension and independent suspension [1]. A suspension with a solid connection between the left and right wheels is called dependent suspension. In a dependent suspension, the movement and vibration of one wheel are transmitted to the opposite wheel directly. The advantages of the solid axle are simple and cheap to manufacture, better vehicle structural, and durability in a high load environment for off-road applications. In an independent suspension on the other hand, the movement of each wheel is independent of the other wheel. The independent suspension allows a wheel to move up and down without affecting the opposite wheel.

Suspensions are also categorized as passive, semi-active, and active systems based on their controllability [2]. The passive suspensions are the most common systems currently used in most vehicles. These systems consist of springs and dampers (shock absorbers) with fixed characteristics which are pre-set and determined according to the design goals and the intended application. The best possible performance is achieved by setting the spring stiffness and the damping; however, the performance is different under various operating conditions because the optimal values are not adjustable. The active suspensions in most cases use hydraulic actuators to generate the desired force. These systems rely entirely on external power to operate the actuators and supply the control forces. Although active components modulate the vertical force reactions of the suspension, they do not alter the kinematics. The semi-active suspensions are a compromise between the active and passive systems. A semi-active suspension may also contain springs and dampers. However, unlike a passive suspension, the properties of these elements (stiffness and damping) are externally controlled and adjusted in real time. The semi-active suspension system offers a desirable performance generally enhanced in the active mode without requiring large power consumption and expensive hardware.

Within the automotive industry, there are trends of evaluating the performance of automotive systems with virtual testing. An important advantage of this kind of simulation consists in the possibility of make virtual measurements for any parameter in any point/area. In this way, the decisions can be made on any design changes without going through physical prototype building and testing. The virtual testing technique is utilized in different applications, such as in vehicle dynamics for modeling tire-roadway interaction, simulating handling maneuvers, braking distance, and assessing durability. With increasing adoption of active and semi-active suspensions, several analytical models of these suspensions have been developed [3-5].

This paper presents the comparative analysis of virtual and experimental proving ground for the performance capabilities of front suspensions in the Family of Medium Tactical Vehicles (FMTV) cargo truck. The front suspension of the current baseline FMTV is a solid axle with leaf springs and shock absorbers. Two other types of suspensions including passive and semi-active suspensions are evaluated in solid and fully independent axle configurations. Virtual proving ground for on- and off-road tests are simulated in the Trucksim environment to include constant radius circular steer, double lane change, sinusoidal steer, washboard road surfaces, and half-round curb strike. Physical proving ground tests are conducted to provide some experimental correlation and validation of the baseline vehicle simulation results. The virtual proving ground is only conducted for the baseline vehicle which has the simplest suspension. For a vehicle with semi-active suspension (air bag and solenoid valve dampers), detail modeling techniques such as multibody dynamic-based simulation are needed. The comprehensive experiments also evaluate the capabilities of various suspensions which have been considered in future FMTV design for mobility performance improvement.

BASELINE VEHICLE AND TEST SCHEMES

The baseline vehicle in this study is M1083A1 (full-time 6x6 drive, 5-ton) FMTV cargo trucks. A solid axle, passive front suspension of the current baseline FMTV has parabolic-tapered leaf springs with hydraulic shock absorbers and stabilizer bar, as shown in Fig. 1 [6]. The truck is ballasted with 4,536 kg (10,000 lbs) with the payload CG located 610 mm (24 inch) above the center of the cargo bed.

Vehicle Modeling and Simulation

The baseline vehicle model is developed in the commercial off-the-shelf nonlinear vehicle dynamics simulation software, Trucksim. The software provides a series of data inputs which contain both graphical and numerical data for the characteristics of the truck size and shape, sprung/unsprung mass, aerodynamics datasets, suspension and tire properties, and vehicle functions (powertrain, steering, and brake), vehicle-roadway interaction, and other factors [7]. System-level behavior is predicted using s high-order mathematical model that solves the nonlinear ordinary differential equations associated with the multibody physics of a vehicle.

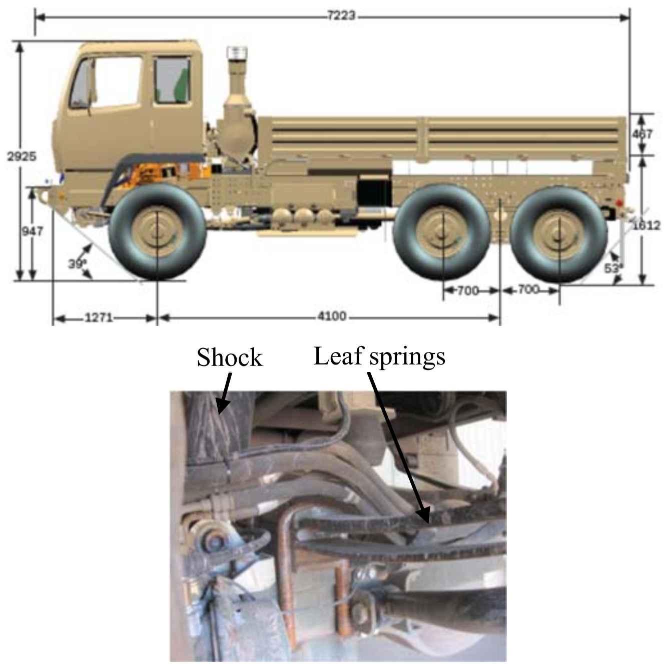

Figure 1: M1083A1 6x6 5-ton baseline truck [6].

On-Road Tests

The purposes of on-road tests are to evaluate suspensions in handling characteristics and response under constant radius circular steer, double lane change, sinusoidal steer, ramp steer, and braking distance. The constant radius circular steer testing is to determine suspension performance at various speeds along a constant turning radius. The vehicle is driven around a circular path with a diameter of 73.8 m measured to the centerline of the vehicle. Counter clockwise tests are conducted at incremental speeds from 16.1 kph – kilometer per hour to the maximum speed where wheel lift is observed. Double lane change testing is conducted to assess the dynamic lateral response by determining the maximum velocity at which the vehicles could successfully traverse the course without wheel lift, knocking cones down, or loss of control. Average vehicle velocity, maximum lateral acceleration, roll angle and yaw rate overshoot are assessed in accordance with North Atlantic Treaty Organization (NATO) AVTP03-160W [8]. The handling characteristics and response of the vehicles are evaluated during a sinusoidal maneuver through a range of lateral accelerations from 0.2g to maximum. The simulation is evaluated at the highest acceleration without wheel lift or outrigger touch.

Ramp steer testing is conducted at 40.2 and 45.1 kph while a counter clockwise steering wheel input of 10 degrees per second is applied until wheel lift occurred. The lateral acceleration, roll angle, and steering wheel angle are measured for each vehicle. Braking performance is evaluated to assess the ability of the vehicle to stop in a controlled manner at 32.2, 48.3 and 80.5 kph based on the guidelines of Federal Motor Vehicle Safety Standard (FMVSS) 121 [9]. The FMVSS121 stopping distance requirements of 23.8 m and 65.8 m are used for the 48.3→0 kph and 80.5→0 kph cases respectively. The 32.2→0 kph case uses 9.75 m requirement from ATPD2131R [10].

Off-Road Tests

The purposes of off-road tests are to evaluate ride quality, shock absorption capability, and side slope grade ability. The ride quality testing is conducted on 25.4 and 50.8 mm Root Means Square (RMS) washboard concrete courses. The ride quality is determined as the maximum vehicle speed at which the absorbed power to the base of the driver's spinal column reached the 6W ride threshold set by ATPD2131R guidelines. Testing for each vehicle speed begins below 16.1 kph and below 8.05 kph on the 25.4 and 50.8 mm profiled course respectively, and then increases incrementally by 1.61 or 3.22 kph until 6W is achieved at the driver seat pad. Additional runs over 6W are conducted to further characterize ride quality near the threshold range. The shock energy absorption test is to determine the vehicle speed at which vertical acceleration at the driver seat foundation exceeds 2.5g. This test is conducted on 203 and 254 mm half-round of single curb strike. Testing for each vehicle begins at low speed, 9.66 kph for 203 mm half-round and 6.44 kph for 254 mm half-round. The vehicle is then increased in 1.61 or 3.22 kph increments until 2.5g vertical acceleration is achieved.

VIRTUAL PROVING GROUND RESULTS

Once the vehicle and testing road parameters are set, the vehicle mathematical model is run through the Trucksim program. After the mathematical model has finished computation, the post processer displays an animation of the truck during simulation and graphical results of simulated parameters on selected components. Figures 2-5 show examples of the animation events of the baseline vehicle in double-lane change, sinusoidal steering, half-round curb strike, and off-road driving.

The simulation result of constant radius circular steer indicates a maximum speed of 43.32 kph can be reached without vehicle roll-over as the vehicle driven around a 73.8 m diameter circular path. The vehicle roll angle versus lateral acceleration is shown in Fig. 6 where the average of lateral acceleration is 0.299g. The vehicle roll angle and roll rate in double lane change simulation is shown in Fig. 7. The maximum roll rate and roll angle is 32.13°/sec and 12.06° respectively in double lane change driving. In sinusoidal steer maneuver, the tire lateral force is shown in Fig. 8 where tire 3 has 15.4 kN maximum force (referring to Fig. 3 for tire numbering). Figure 9 shows steering wheel angle and vehicle roll angle. Figure 10 also shows the relationship between vehicle roll angle and lateral acceleration. The maximum roll angle reaches 11.68° with 0.416g lateral acceleration. A simulation result of braking performance is shown in Fig. 11. The result indicates that a 53.3 m of braking distance is needed for a vehicle travelling with 80.5 kph initial speed. A maximum of 0.63g longitudinal deceleration is occurred. An oscillation of 0.01g, as shown in Fig. 11, is mainly due to numerical convergence in the simulation.

Figure 2: Simulation of double-lane change.

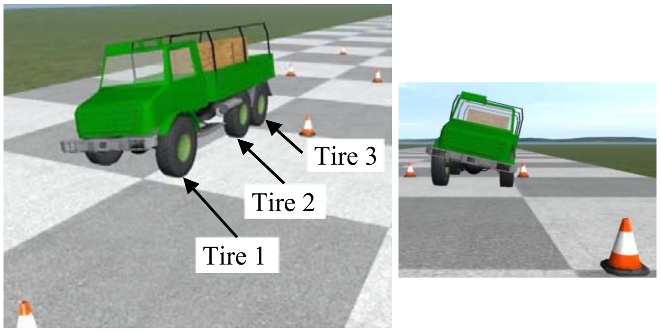

Figure 3: Simulation of sinusoidal steering.

Figure 4: Simulation of half-round single curb strike.

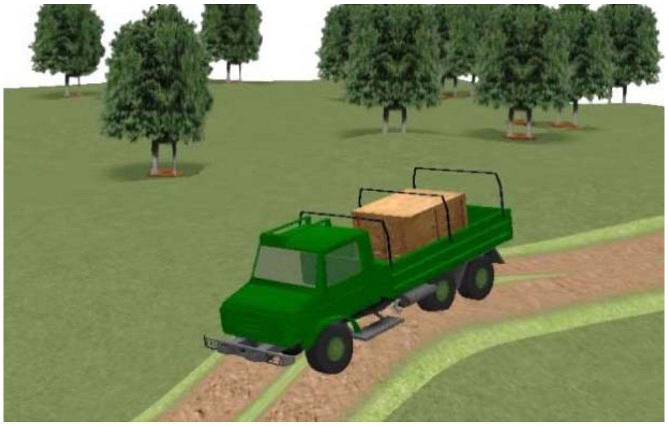

Figure 5: Simulation of off-road driving.

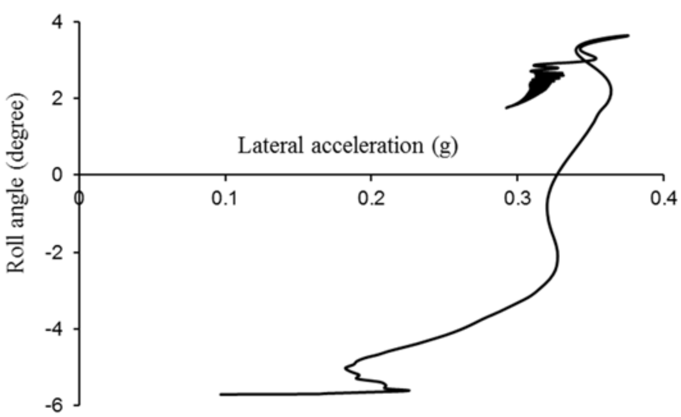

Figure 6: Vehicle roll angle versus lateral acceleration in a circular path driving.

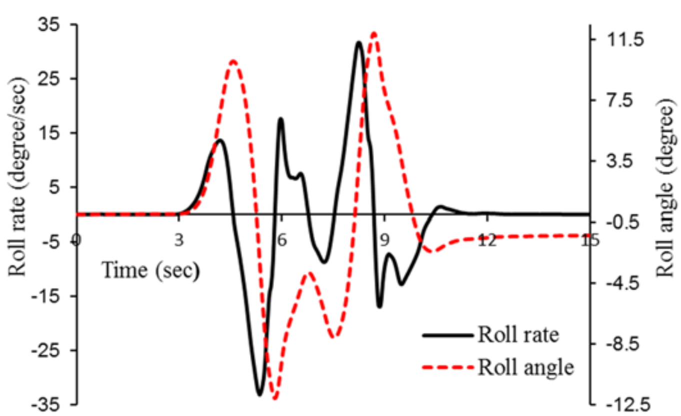

Figure 7: Vehicle roll rate and roll angle in double lane change.

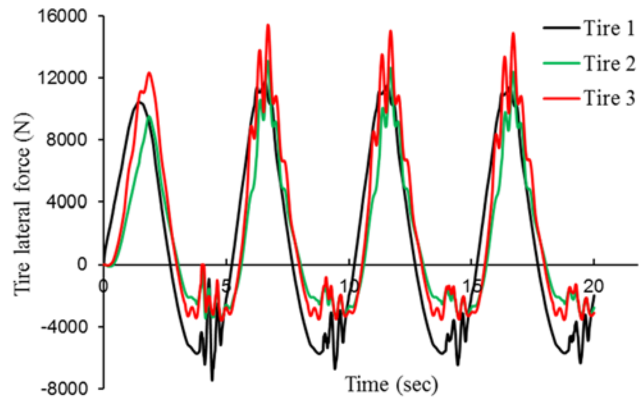

Figure 8: Tire lateral force in sinusoidal steering.

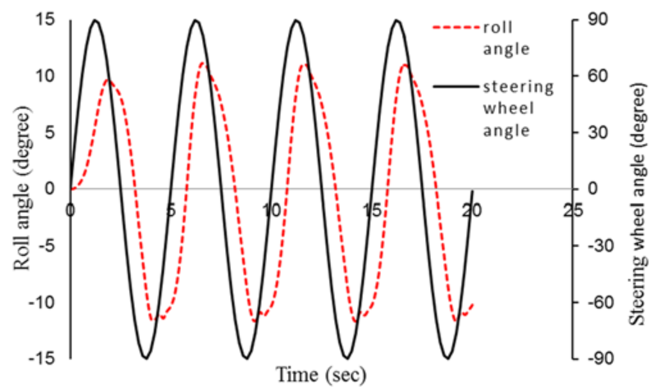

Figure 9: Steering wheel angle and vehicle roll angle in sinusoidal steering.

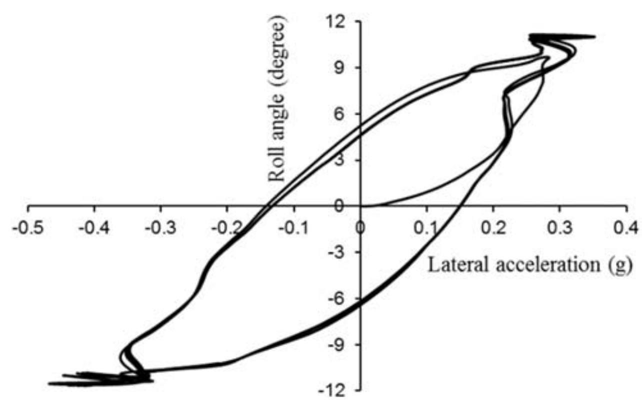

Figure 10: Vehicle roll angle and corresponding lateral acceleration in sinusoidal steering.

One of the off-road simulation results shown in Fig. 12 is a 203 mm half-round single curb strike. Over the curb, vehicle speed reaches 17 kph and a 2.56g maximum vertical acceleration occurs. The vehicle pitch angle is up to 5.28°.

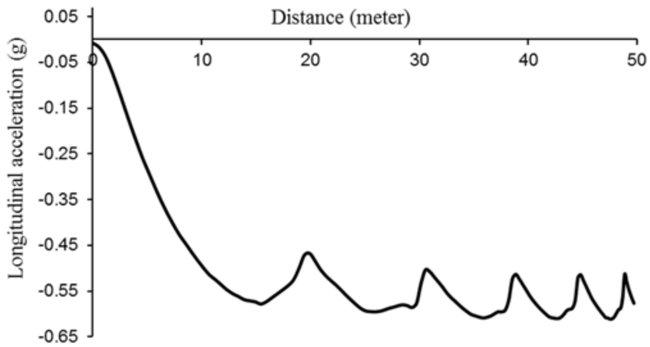

Figure 11: Simulation result of braking distance with 80.5 kph initial speed.

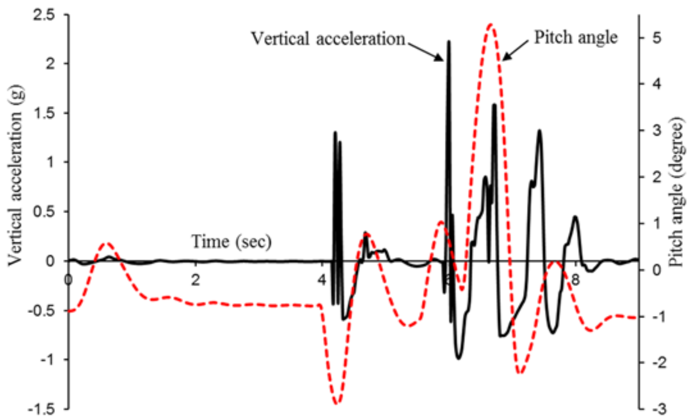

Figure 12: Vehicle vertical acceleration and pitch angle within 203 mm half-round single curb strike.

EXPERIMENTAL RESULTS

Table 1 summarizes the experimental tests include: the baseline (solid axle with leaf springs and shock absorber), Meritor independent suspension with coil springs over shocks [11], and ADVS [12]. The ADVS utilized air bag suspension, Solenoid Valve (SV) dampers and leaf springs. The baseline vehicles have the leaf springs consisting of four layers of leaves stacked on top of each other. The Meritor is independent suspensions without leaf springs. All trucks with suspension variants are ballasted with payload installed in the cargo bed 4,536 kg with the payload CG located 610 mm above the center of the cargo bed. Two photos of experimental tests, Figure 13 (sinusoidal steering testing) and Fig. 14 (203 mm half-round single curb strike), respectively corresponds to the simulation events shown in Fig. 3 and 4.

Double lane change testing is conducted by determining the maximum velocity at which the vehicles could successfully traverse the course. The ADVS vehicle has a maximum average speed of 67.9 kph that is 15.8 % lower than the baseline vehicle. The ADVS suspension also records a lower roll angle, roll rate, and yaw rate overshoot than the baseline vehicle. In sinusoidal steering tests, the Meritor and ADVS suspensions are all tested at higher lateral accelerations without wheel lift occurring, but they could not maintain a speed within the testing parameters due to excessive tire scrubbing. Each of the other suspension variants record maximum lateral accelerations between 0.46 and 0.49g. Table 2 summarizes the test results of constant radius circular steering, double lane change, sinusoidal steering, and ride quality during half-round curb strike. The ADVS proved comparable to the baseline suspension in roll angle, but produces 1.6° more roll angle than the Meritor suspension in sinusoidal steering. In measuring the braking distance, all vehicles achieve the FMVSS121 requirements for 48.3 m and 80.5 kph stopping distances except the ADVS. The baseline vehicle also fails to meet either the ATPD2131R or FMVSS121 requirement for 32.2 kph stopping distance.

Table 1. Tested vehicles with variant suspensions

	Axle type	Front springs	Rear springs	Shock absorber
Baseline	solid	leaf springs	leaf springs	passive
Meritor with coil springs	solid	coil-over springs	coil-over springs	passive
ADVS live beam	independent	leaf springs and air springs	air springs	SV damper, semi-active

Figure 13: Sinusoidal steer testing.

Figure 14: A 203 mm half-round single curb strike.

For the ride quality test results, the suspension shock absorbed power is plotted as a function of vehicle speed passing through the 25.4 and 50.8 mm RMS washboard road surface. All vehicle suspension variants exceed the 27.4 kph with 6W ride threshold for 25.4 mm RMS. The ADVS suspension shows degraded ride quality performance, reaching a 6W ride speed of only 28.5 kph that is 4.8% and 14.5% less than the baseline and Meritor respectively. Figure 15 shows the absorbed power relative to vehicle speed over the 50.8 mm RMS course. As shown in Fig. 15, the ADVS suspension has a 25% increase in 6W ride speed compared to the baseline, but a 19% speed reduction compared to the Meritor-coil suspension.

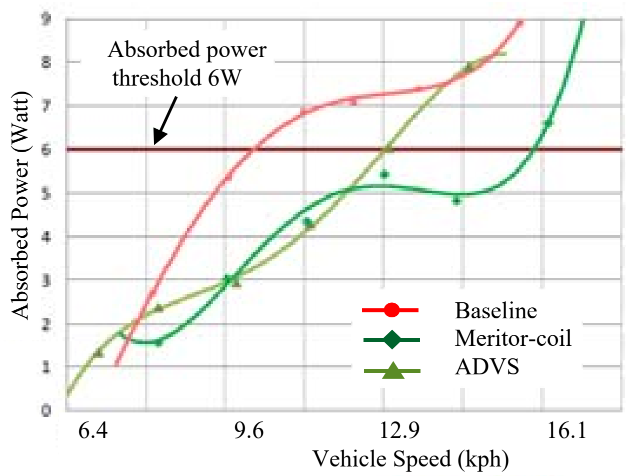

Figure 15: Absorbed power over 50.8 mm RMS course.

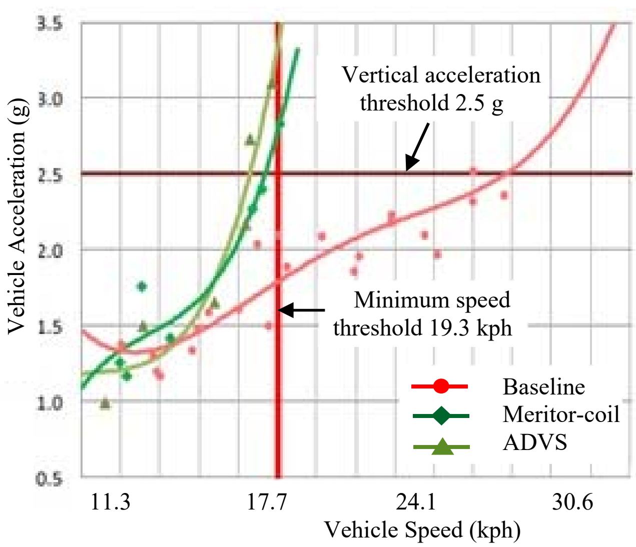

Figure 16: Vertical acceleration comparison in the 203 mm half-round single curb strike.

The half-round single curb strike is used to determine the shock absorption capability. During the 203 and 254 mm half-round single curb strike, the vehicle speed is recorded at which vertical acceleration at the driver seat foundation exceeds 2.5g. Figure 16 shows each vehicle speed relative to the vertical acceleration at the driver seat frame under the tests. In the 203 mm half-round test results, the ADVS suspension has a 3% speed reduction over the baseline. Both ADVS and baseline suspensions fail to meet the ATPD2131R ride quality requirement of 19.3 kph over the 203 mm half-round curb strike. The ADVS suspension shows no increase over the baseline in the 254 half-round curb strike.

Table 2. Summary of test results

Constant radius circular steering					
	speed (kph)	avg. lateral accel. (g)	avg. roll angle (deg)	steering wheel angle (deg)	gradient steering wheel (deg/g)
Baseline	43.3	0.37	7.3	273	190
Meritor-coil	43.1	0.41	6.2	279	117
ADVS	43.60	0.41	9.7	127	119
Double lane change - data at max. vehicle speed					
	speed (kph)	max. lateral accel. (g)	max. roll angle (deg)	max. roll rate (deg/s)	yaw rate overshoot (deg/s)
Baseline	80.6	0.39	8.6	25.5	8.2
Meritor-coil	79.8	0.42	8.7	28.2	6.5
ADVS	67.9	0.44	8.0	13.4	6.1
Sinusoidal steering - data at max. vehicle speed					
	mean lateral accel. (g)	mean roll angle (deg)	mean roll rate (deg/s)	mean yaw rate (deg/s)	mean steering wheel angle (deg)
Baseline	0.41	8.3	21.5	19.1	192
Meritor-coil	0.45	7.2	20.0	19.4	225
ADVS	0.46	8.8	18.8	26.6	760
ATPD2131R 3.2.1.14 ride quality test					
	203 mm half-round		254 mm half-round		
	Speed (kph)	Comparison to baseline	Speed (kph)	Comparison to baseline	
Baseline	18.8		17.7		
Meritor-coil	28.6	52%	17.4	-1.7%	
ADVS	18.2	-3%	17.7	0%	

CONCLUSIONS

This paper presents the virtual proving ground testing and experimental validation for the performance capabilities of front suspensions in medium-duty truck. Virtual proving ground for on- and off-road tests are simulated in the Trucksim environment to include constant radius circular steer, double lane change, sinusoidal steer, washboard road surfaces, and half-round curb strike. Physical proving ground tests are conducted to provide some experimental correlation and validation of the baseline vehicle simulation results. The on-road simulation results have acceptable correlation with the proving ground experimental results as illustrated in Table 3. The virtual proving ground is only conducted for the baseline vehicle which has the simplest suspension. For a vehicle with semi-active suspension (air bag and SV dampers), more detail modeling techniques such as multibody dynamic-based simulation are needed.

The front suspension of the current baseline FMTV is a solid axle with leaf springs and shock absorbers. Two other types of suspension systems including passive and semi-active suspensions are evaluated in solid axle and fully independent configurations. The semi-active suspension has a 15.8% speed decrease in the double lane change comparing to the baseline. Off-road performance testing for the semi-active suspension shows a 14.5% and 19% reduction of vehicle speed on 25.4 and 50.8 mm RMS washboard respectively compared to the Meritor-coil suspension, and comparable performance in both 203 mm and 254 mm half-round curb strike. The ADVS semi-active suspension had degraded performance compared to the baseline in several tests, which did not meet the expectation of a typical semi-active suspension. This phenomenon might be due to that ADVS active shock dampers were originally designed for M1078 FMTV cargo trucks (4x4 drive, 2.5-ton). The comprehensive experiments also evaluate the capabilities of various suspensions which have been considered in future FMTV design for mobility performance improvement.

Table 3. Comparisons between virtual tests and experiments

		Virtual test	Experimental result
Constant radius circular steer	max. speed (kph)	43.3	43.3
	lateral accel. (g)	0.30	0.37
Double lane change	roll angle (deg)	12.1	8.6
	roll rate (deg/s)	32.1	25.5
	lateral accel. (g)	0.42	0.39
Sinusoidal steer maneuver	roll angle (deg)	11.7	8.3
	lateral accel. (g)	0.42	0.41
Braking distance (m) Vehicle at 80.5 kph initial speed		53.3	50
Vehicle speed (kph) at a 203 mm half-round single curb strike (driver seat acceleration)		17.2 (2.56 g)	18.8 (2.50 g)

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